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DESCRIPTION

REFRIGERANT COMPRESSOR

TECHNICAL FIELD

The present invention relates to an improvement in efficiency of a hermetic compressor used in freezer-refrigerators, and the like.

BACKGROUND ART

Recently, an improvement in efficiency of a hermetic compressor used in freezer-refrigerators, and the like, has been strongly demanded. In a conventional hermetic compressor, the suction efficiency is increased by, for example, providing a valve device of a compression part with two holes, and the compression efficiency is improved. Such a compressor is disclosed in, for example, Japanese Patent Unexamined Publication No. H3-175174. Hereinafter, an example of a conventional hermetic compressor is described with reference to drawings.

FIG. 6 is a sectional view showing a conventional refrigerant compressor; and FIG. 7 is an exploded perspective view showing a valve of a conventional refrigerant compressor. To a hermetic container 51, an outlet port 52A that is one end of a suction tube 52 is connected. Another end of the suction tube 52 is connected to a piping at the low-pressure side of a refrigerating cycle (not shown). A motor 53 includes a stator 54 and a rotor 55, so as to drive a compression part 56. Furthermore, refrigerating machine oil 57 is preserved in a bottom portion of the hermetic container 51. A coil spring 58 elastically supports the motor 53 and the compression part 56.

The compression part 56 includes a cylinder head 61, a cylinder block

62, a valve plate 64, a suction reed valve 67, a piston 68, a connecting rod 70 and a suction muffler 30. The cylinder head 61 forms a suction space 61A and a discharge space 61B. The cylinder block 62 contains a cylinder 63. The valve plate 64 has two suction holes 65 and two discharge holes 66. The suction reed valve (hereinafter, referred to as "valve") 67 has a deformation part 67A. The connecting rod 70 is linked to an eccentric part 69A of a crank shaft 69. The suction muffler 30 communicates to a suction space 61A via a communicating tube 30A and to the suction hole 65 of the valve plate 64, and sucks a refrigerant gas from an inlet port 30B.

Hereinafter, an operation of the refrigerant compressor configured as mentioned above is described. Firstly, the motor 53 drives the compression part 56, so that the piston 68 reciprocates in the cylinder 63. A low temperature and low pressure refrigerant gas returning from an external refrigerating cycle (not shown) is firstly sucked into the hermetic container 51 from the suction tube 52. The refrigerant gas is further sucked from the inlet port 30B of the suction muffler 30 and passes through the suction hole 65 via the communicating tube 30A. During the suction stroke, by flexing the deformation part 67A of the valve 67, the refrigerant gas opens the valve 67 and is led to the cylinder 63. During the compression stroke, the valve 67 is closed. Thus, the refrigerant gas is compressed to high temperature and high pressure, passes from the discharge hole 66 through a discharge tube (not shown) and is led to the external refrigerating cycle (not shown) so as to be used for a refrigerating operation.

At this time, the valve 67 is so designed as to have a natural frequency for carrying out an opening and closing operation with good timing in accordance with a low-speed operation frequency. Therefore, the compressor is capable of operation with a reduced suction loss and a high volumetric

efficiency.

However, when a low-speed operation frequency is changed to a high-speed operation frequency due to the change in cooling load conditions, the timing of opening and closing operation determined by the natural frequency of the valve 67 is off. At this time, even when the pressure in the cylinder 63 exceeds the pressure in the suction space 61A of the cylinder head 61, the valve 67 does not complete a closing operation. Consequently, delayed closing occurs, causing a backflow of a refrigerant gas, so that the volumetric efficiency is lowered and the refrigerating capacity and the refrigerating efficiency are lowered.

As a solution to reduce the backflow of a refrigerant gas due to delayed closing of the valve 67, designing the valve 67 to have a high natural frequency in accordance with the high-speed operation is considered. In this case, since the spring constant of the deformation part 67A is increased, an amount of flexure is reduced, so that the suction loss is increased and accordingly the refrigerating capacity and the refrigerating efficiency are lowered.

SUMMARY OF THE INVENTION

The refrigerant compressor of the present invention includes a piston, a cylinder and a valve plate. The valve plate is provided at an opening end of the cylinder and includes a plurality of suction holes. The refrigerant compressor of the present invention further includes a plurality of suction reed valves, which is provided between the opening end of the cylinder and the valve plate, for opening and closing the plurality of suction holes, respectively. At least one of the suction reed valves has a natural frequency different from that of the other reed valves. With this configuration, even when an operation frequency is changed, delayed closing of the suction reed valve and reduction in

an amount of flexure are prevented.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view showing a refrigerant compressor in an embodiment of the preset invention.

FIG. 2 is a front view showing a suction reed valve of the refrigerant compressor shown in FIG. 1.

FIG. 3 is a sectional view showing a cylinder head part of the refrigerant compressor shown in FIG. 1.

FIG. 4 is a graph showing a pressure in a cylinder and an amount of flexure of a reed valve in one stroke in a low-speed operation of a refrigerant compressor in an embodiment of the present invention.

FIG. 5 is a graph showing a pressure in a cylinder and an amount of flexure of a reed valve in one stroke in a high-speed operation of a refrigerant compressor in an embodiment of the present invention.

FIG. 6 is a sectional view showing a conventional refrigerant compressor.

FIG. 7 is an exploded perspective view showing a valve of the refrigerant compressor of FIG. 6.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

FIG. 1 is a sectional view showing a refrigerant compressor in an embodiment of the preset invention. FIG. 2 is a front view showing a suction reed valve. FIG. 3 is a sectional view showing a cylinder head part.

To a hermetic container 1, an outlet port 2A that is one end of a suction tube 2 is connected. Another end of the suction tube 2 is connected to a piping at the low-pressure side of a refrigerating cycle (not shown). A motor 3

includes a stator 4 and a rotor 5, and drives a compression part 6. Furthermore, refrigerating machine oil 7 is preserved in a bottom portion of the hermetic container 1. A coil spring 8 elastically supports the motor 3 and the compression part 6.

The compression part 6 includes a cylinder head 101, a cylinder block 12, a valve plate 110, suction reed valves (hereinafter, referred to as "valve") 120A and 120B, a piston 18, a connecting rod 20 and a suction muffler 130. The cylinder head 101 forms a suction space 101A and a discharge space 101B. The cylinder block 12 contains a cylinder 13. The connecting rod 20 is linked to an eccentric part 19A of a crank shaft 19. The suction muffler 130 communicates to a suction space 101A via a communicating tube 130A and to suction holes 112A and 112B of the valve plate 110, and sucks a refrigerant gas from an inlet port 130B.

The valve plate 110 has the suction holes 112A and 112B and discharge holes (not shown). The suction holes 112A and 112B are respectively inclined from opening portions 114A and 114B of the valve plate 110 at the side of the cylinder 13 to opening portions 114C and 114D of the valve plate 110 at the side of cylinder head 101 in the direction in which a distance between the suction holes is reduced. The valves 120A and 120B have deformation parts 122A and 122B having different lengths each other, respectively. Since the deformation part 122A is longer than the deformation part 122B, the spring constant of the valve 120A is smaller, and the valve 120A has a lower natural frequency than that of the valve 120B. Furthermore, the shapes of the valves 120A and 120B are asymmetric with respect to centerlines 124A and 124B of the deformation parts 122A and 122B. Positions of center points of the suction holes 112A and 112B correspond to points 126A and 126B of the valves 120A and 120B, respectively.

Seal parts 128A and 128B seal the suction holes 112A and 112B provided on the valve plate 110.

Hereinafter, an operation of the refrigerant compressor configured as mentioned above is described. FIG. 4 is a graph showing a pressure in a cylinder and an amount of flexure of a reed valve in one stroke in a low-speed operation of a refrigerant compressor in an embodiment of the present invention. FIG. 5 is a graph showing a pressure in a cylinder and an amount of flexure of a reed valve in one stroke in a high-speed operation of a refrigerant compressor in an embodiment of the present invention.

The motor 3 drives the compression part 6, so that the piston 18 reciprocates in the cylinder 13. A low temperature and low pressure refrigerant gas returning from an external refrigerating cycle (not shown) is firstly sucked into the hermetic container 1 from the suction tube 2. The refrigerant gas is further sucked from the inlet port 130B of the suction muffler 130 and passes through the suction holes 112A and 112B via the communicating tube 130A. During the suction stroke, by flexing the deformation parts 122A and 122B of the valves 120A and 120B, the refrigerant gas opens the valve 120A and 120B and is led to the cylinder 13. During the compression stroke, the valves 120A and 120B are closed, and the refrigerant gas is compressed to high temperature and high pressure, passes from the discharge hole through a discharge tube (not shown) and is led to the external refrigerating cycle to be used for a refrigerating operation.

When the piston 18 reciprocates in the cylinder 13, in the suction stroke, the piston 18 shifts to the side of a bottom dead center. In a low-speed operation, in this suction stroke, gas pressure load, which is generated by differential pressure when the pressure 140 in the cylinder 13 becomes lower than the pressure in the suction space 101A of the cylinder head 101, acts on

the valves 120A and 120B. At this time, at a point 140A, the suction reed valves 120A and 120B start to open, and the refrigerant gas is sucked into the cylinder 13. The point 140A means a point at which the gas pressure load generated by differential pressure becomes larger than a resultant force of flexure load of the valves 120A-120B and an adhesion force with the viscosity of refrigerating machine oil at the seal parts of the valves 120A-120B.

Furthermore, in the compression stroke, the valves 120A and 120B are closed at a point 140B at which the pressure in the cylinder 13 exceeds the pressure in the suction space 101A of the cylinder head 101, and suction of the refrigerant gas from the suction muffler 130 is completed.

Between the point 140A and the point 140B, the valve 120A repeats opening and closing operations 150A twice at a natural frequency in primary deformation mode while flexing the deformation part 122A. Since the valve 120A is selected to have a natural frequency corresponding to a low-speed operation frequency, the valve 120A completes closing substantially with the same timing as the point 140B. Furthermore, since the spring constant of the valve 120A is small, even under conditions that the flow rate of sucked gas is slow during a low-speed operation, a suction loss due to the shortage of an amount of flexure is not increased.

Furthermore, the valve 120B has a natural frequency and a spring constant higher than those of the valve 120A, and repeats opening and closing operations 150B four times between the point 140A and the point 140B. At this time, the valve 120B opens widely with a certain amount of flexure according to the circulation amount of refrigerant at the first to third opening and closing operations 150B. At the fourth opening and closing operation, since in the compression stroke, the differential pressure between the pressure in the cylinder 13 and the pressure in the suction space 101A of the cylinder

head 101 is extremely small. At this time, the refrigerant gas flows in the suction hole 112A of the valve 120A that flexes more largely. Consequently, the amount of a refrigerant gas flowing in the suction hole 112B of the valve 120B becomes very small and the dynamic pressure by the flow of the refrigerant gas is reduced. That is to say, the valve 120B hardly flexes and it completes opening and closing operation near the point 141B.

Therefore, backflow of the refrigerant gas due to delayed closing of the valves 120A and 120B is prevented, and the increase in suction loss due to the shortage of the amount of flexure in the suction stroke is also prevented. As a result, the volumetric efficiency is increased.

Furthermore, in a high-speed operation, between the point 141A and the point 141B, the valve 120B repeats opening and closing operations 151B three times and flexes with a certain amount of flexure according to the circulation amount of refrigerant, and then completes closing with good timing. The point 141A means a point at which the pressure in the cylinder 13 becomes lower than the pressure in the suction space 101A of the cylinder head 101. Furthermore, the point 141B means a point at which the pressure in the cylinder 13 exceeds the pressure in the suction space 101A of the cylinder head 101.

The valve 120A opens widely in a certain amount of flexure according to the circulation amount of refrigerant at the first opening and closing operation 151A. On the other hand, in the second opening and closing operation, since in a compression stroke, the differential pressure between the pressure in the cylinder 13 and the pressure in the suction space 101A of the cylinder head 101 is extremely small. Therefore, the refrigerant gas passes through the suction hole 112B of the valve 120B that flexes more largely. Consequently, the valve 120A hardly flexes and it completes the opening and closing operation near the

point 141B.

Therefore, also in a high-speed operation, delayed closing or shortage of the amount of flexure of the valves 120A and 120B do not occur, so that the refrigerant gas is sucked into the cylinder 13 efficiently. Consequently, in a case where the operation frequency is changed, the refrigerating capacity and the compression efficiency of the compressor are increased.

Furthermore, the shapes of the valves 120A and 120B are asymmetric with respect to the centerlines 124A and 124B of the deformation parts 122A and 122B. Therefore, working points 126A and 126B of the gas pressure load that act on the valves 120A and 120B deviate from centerlines 124A and 124B of the flexing deformation of the valves 120A and 120B. Thus, the valves 120A and 120B start to open with torsion deformation. That is to say, torsional moment due to gas pressure load acts on the valves 120A and 120B. Consequently, on one side of circular seal parts 128A and 128B of the valves 120A and 120B, force for peeling off the adhesion portion with a viscosity of the refrigerating machine oil 7 is concentrated, thus making it easy to open the valves 120A and 120B. Accordingly, in the suction stroke, the valves 120A and 120B start to open earlier. Therefore, the refrigerant gas is sucked into the cylinder 13 efficiently, and the refrigerating capacity and the compression efficiency of the compressor are increased. Note here that in FIG. 2, both of the shapes of the valves 120A and 120B are asymmetric with respect to the centerlines 124A and 124B of the deformation parts 122A and 122B. However, they may be configured so that only one of the shapes is asymmetric.

A refrigerant gas inside the hermetic container 1 passes through the suction space 101A in a high-temperature cylinder head 101 via the suction muffler 130 and is sucked into the cylinder 13 from the suction holes 112A and 112B provided on the valve plate 110. Herein, the refrigerant gas inside the

cylinder 13 is in a high temperature state of about 100°C by a compression action and is discharged to a discharge space 101B of the cylinder head 101. Thus, the cylinder head 101 is heated to high temperature state of about 80°C.

In this case, as an interval between the two suction holes 112A and 112B of the suction space 101A in the cylinder head 101, at least a distance corresponding to a sum of the widths of the seal part 128A and the seal part 128B is necessary. Herein, as shown in FIG. 3, in a case where the suction holes 112A and 112B are inclined, it is not necessary to consider the widths of the seal part 128A and the seal part 128B, and it is possible to greatly reduce the interval between the suction holes 112A and 112B. Thus, the suction space 101A in the cylinder head 101 can be configured to have a reduced volume and heat receiving area, so that the thermal transmission to the flowing refrigerant gas is reduced. Accordingly, the temperature of the refrigerant can be kept low, the density of the gas refrigerant increases and the circulation amount of the refrigerant increases. Consequently, the refrigerating capacity and the compression efficiency of the compressor increase. Note here that in FIG. 3, both of the suction holes 112A and 112B are inclined, but only one of them may be inclined.

Note here that in the embodiment, the number of the valves 120A and 120B is two. However, when the number is three or more, the same effect can be obtained.

Furthermore, in the embodiment, the natural frequency is changed by varying the lengths of the valves 120A and 120B. However, the same effect can be obtained even when the natural frequency is changed by varying the widths or shapes of the valves 120A and 120B.

Furthermore, in the embodiment, the case where the number of opening and closing operations in one stroke of the valve 120A and 120B is two to four is

described, however the same effect can be obtained when the number is one or more.

INDUSTRIAL APPLICABILITY

A refrigerant compressor of the present invention includes a piston, a cylinder and a valve plate. The valve plate is provided at an opening end of the cylinder and has a plurality of suction holes. The refrigerant compressor of the present invention further includes a plurality of suction reed valves provided between the opening end of the cylinder and the valve plate, and opens and closes the plurality of suction holes, respectively. At least one of the suction reed valves has a natural frequency different from that of the other reed valves. With this configuration, since refrigerating capacity and compression efficiency of the refrigerant compressor can be increased, the refrigerant compressor can be used for air conditioners, freezer-refrigerators, and the like.